Research on Convective Heat Transfer and Mass Transfer of the Evaporator in Micro/Mini-Channel

Jianmin Su
Master
Beijing Institute of Civil Engineering and Architecture
Beijing P.R.China,100044
sjm19820623@sohu.com

Junming Li
Professor
Tsinghua University
Beijing P.R.China,100084
lijm@tsinghua.edu.cn

Abstract: With the development of science and technology, various heating and cooling equipment have a development trend of micromation. Micro-fabrication processes make it possible to conduct research on condensation heat transfer in micro-channels. Based on the reviewers on the present household air conditioners, the potential requirements for new heat transfer enhancement used for household air conditioners are discussed. Investigations on condensation and boiling of refrigerants in mini/micro channels have indicated that the evaporator and condenser of air conditioner would be more efficient and more compact by using microchannels, and hence it could improve the coefficient of performance of air conditioners to meet the new energy conversion standards in China. The relationship between condensation heat transfer of refrigerants and surface physical characteristics of the evaporator are pointed out and analyzed in order to achieving the corresponding heat transfer coefficients.

Key words: micro/mini-channel; evaporator; convective heat transfer coefficients

1. INTRODUCE

With the development of science and technology, various heating and cooling equipments have a development trend of micromation. Micro-fabrication processes make it possible to research on condensation heat transfer in micro-channels. The studies of compact microchannels evaporators in the automotive , aerospace and cryogenic industries have been a hot investmental topic. But at present the design theory about compact microchannels evaporators is not very mature. Compact microchannels evaporators is studied and discussed in this paper, based on heat transfer in microchannel and mechanism of boiling heat transfer.

2. CALCULATION OF EVAPORATOR

This is an ideal slab microchannel heat exchanger. Fig. 1 presents a schematic diagram of the folded multi-louvered fin geometry and Table 1 shows the specifications of the slab microchannel heat exchanger.

2.1 Air-Side Heat Transfer Coefficient

Kim and Bullard[1,2] compared their air-side heat transfer correlation with test data from several other microchannel heat exchanger geometries. They reported that their correlation predicted the data well. Therefore, their correlation was adopted in the model.

For dry surfaces, Kim and Bullard developed the j and f correlations for Re_{Lp}:100–600 and Fp/Lp<1 with root-mean-square (RMS) errors of 14.5 and 7%, respectively:

$$j=Re_{\mathrm{Lp}}^{-0.487}\left(\frac{L_\alpha}{90}\right)^{0.257}\left(\frac{F_\mathrm{p}}{L_\mathrm{p}}\right)^{-0.13}\left(\frac{H}{L_\mathrm{p}}\right)^{-0.29}\left(\frac{F_\mathrm{d}}{L_\mathrm{p}}\right)^{-0.235}\left(\frac{L_\mathrm{l}}{L_\mathrm{p}}\right)^{0.68}\left(\frac{T_\mathrm{p}}{L_\mathrm{p}}\right)^{-0.279}\left(\frac{\delta_\mathrm{f}}{L_\mathrm{p}}\right)^{-0.05}$$

$$f=Re_{\mathrm{Lp}}^{-0.781}\left(\frac{L_\alpha}{90}\right)^{0.444}\left(\frac{F_\mathrm{p}}{L_\mathrm{p}}\right)^{-1.682}\left(\frac{H}{L_\mathrm{p}}\right)^{-1.22}\left(\frac{F_\mathrm{d}}{L_\mathrm{p}}\right)^{0.818}\left(\frac{L_\mathrm{l}}{L_\mathrm{p}}\right)^{1.97}$$

Where $Re_{Lp}=V_cL_p/\nu$, V_c is air velocity through minimum free-flow area. ν is kinematic viscosity. And j is defined as:

$$j=\frac{h_\mathrm{o}}{\rho_\mathrm{m}V_\mathrm{c}c_{\mathrm{p,o}}}Pr_\mathrm{o}^{2/3}$$

If air velocity V_c is 2.5m/s, air inlet temperature t_1 is 26 centigrade, air outlet temperature t_2 is 16 centigrade, we can gain the physical characteristic of R22.

ν =15.154×10^{-6}m^2/s , Pr_o=0.7082 , Cp_o=1005J/(kg*k)，ρ_m=1.201kg/m^3

For $Re_{Lp}=V_cL_p/\nu$ =2.5×1.7×10^{-3}/(15.154×10^{-6})=280

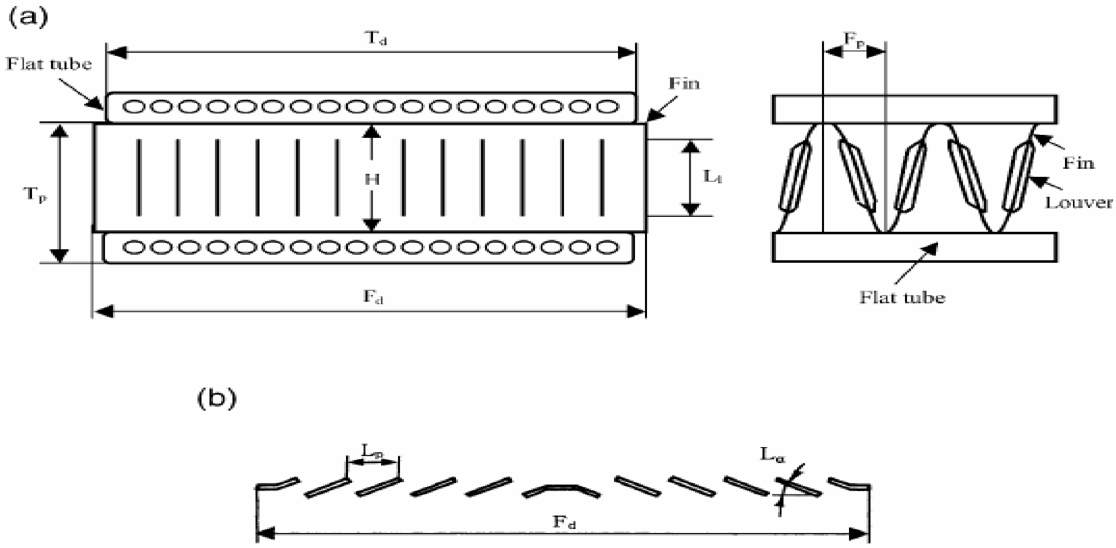

Fig. 1 Schematic illustration of folded multi-louvered fin geometry: (a) definition of geometric parameters and (b)cross-section of louvered fin geometry

Tab.1 Specifications of the ideal heat exchanger

T_d	Tube depth (mm)	24.2×2
H	Fin height (mm)	8
T_p	Fin height and slab thickness	10
F_d	Flow depth per slab (mm)	24.2
L_l	Louver length (mm)	7
F_p	Fin pitch (mm)	1.4
L_p	Louver pitch (mm)	1.7
L_a	Louver angle（°）	27
δ_f	Fin thickness (mm)	0.1
	Port diameter (mm)	1

2.2 Refrigerant Side Heat Transfer Coefficient

and F_p / L_p=0.82<1，so the calculation of j is 0.0304. $h_o=j*\rho_m*V_c*Cp_o*Pr_o^{-2/3}$=0.0304×1.201×2.5×1005×0.7082$^{-2/3}$=115w/(m^2*k)Tran[3] supplied a boiling heat transfer correlation, based on the boiling heat transfer experiment in circular and rectangular microchannel. They found that convective heat transfer and nucleate boiling heat transfer are the primary mode of heat transfer in circular and rectangular microchannel. But these two modes of heat transfer can be changed each other under definite conditions. They also found that if the superheat degree of wall is more than 2.75 centigrade, the effect of mass flux and vapor quality can be neglected. But if the superheat degree of wall is less than 2.75 centigrade, mass flux and vapor quality affect heat transfer in different degree. They judged that the gist of modes of heat transfer in microchannel is superheat degree of wall. When superheat degree of wall is very small, heat transfer is referable to mass flux and vapor quality. Convective heat transfer is in a dominant position. When superheat degree of wall is very large, heat transfer is referable to heat flux. Nucleate boiling heat transfer is in a dominant position. At last, they revised the heat transfer correlation about conventional channel and supplied a boiling heat transfer correlation about microchannel as follows:

$$h=(8.4\times10^5)(Bo^2We_l)^{0.3}(\rho_l/\rho_v)^{-0.4}$$

$$Bo=q/(i_{fg}*G) \quad We_l=G^2de/(\rho_l*\sigma)$$

In this model, if evaporation temperature t_0 is 5 centigrade, condensation temperature t_k is 40 centigrade, heat flux q is 500 w/m^2, refrigerant mass flux q_m is 300kg/(m^2*s), the physical characteristic of R22 ρ_l is 1267.395 kg/m^3, ρ_v is 25.525 kg/m^3, σ is 0.0112 N/m, i_{fg} is 201160 J/kg, we can get the result that heat transfer coefficient h is 274 w/(m^2*k).

2.3 Heat Transfer Coefficient of Evaporator

Enlarge the area of fin can minish heat resistance and boost up heat transfer. So fin is often used to boost up heat transfer. Given fin and wall is the same material, A_2 is the area of fin, A_1 is the area of wall.

Fin efficiency of the heat exchanger can be described as follows[4]:

$$\eta_f = \frac{th(ml)}{ml}, m = \sqrt{\frac{2h_o}{k_f \delta_f}(1+\frac{\delta_f}{F_d})}, l = \frac{H}{2} - \delta_f$$

According to the size of evaporator, we can get the result that η_f is 94.6%, A_2 is 0.177 m^2, total fin efficiency η is 95%, the coefficient of fin β is 6.3.

Heat transfer coefficient of evaporator can be described as follows:

$$k = \frac{1}{\frac{1}{h_1}\beta + \frac{\delta}{\lambda}\beta + \frac{1}{h_2 \eta}}$$

For microchannel δ is wall thickness, which can be neglected.So we can get the result that heat transfer coefficient of evaporator k is 32 w/(m^2*k).

2.4 Heat Transfer Area of Evaporator

Difference in temperature of evaporator can be described as follows:

$$\theta_m = \frac{t_1 - t_2}{\ln \frac{t_1 - t_e}{t_2 - t_e}}$$

Heat transfer area of evaporator can be described as follows:

$$A = \frac{Q_e}{K \theta_m}$$

When air inlet temperature t_1 is 26 centigrade, air outlet temperature t_2 is 16 centigrade, evaporation temperature t_e is 5 centigrade, Q_e is 2300w, we can get the result that heat transfer area of evaporator is 4.6 m^2.

According to the size of evaporator, we can get the result that weather area is 0.077 m^2, which is reduced by 20% compared to others.

3. IMPACT FACTOR OF EVAPORATOR

3.1 Impact of Air Velocity

When air velocity is changed, air-side heat transfer coefficient is showed in table 2. It shows that air-side heat transfer coefficient can be increased by increasing air velocity.

Tab.2 Results of test

air velocity V_c (m/s)	1.5	2.0	2.5	3.0	3.5
air-side heat transfer coefficient h_o	89	103	115	127	137

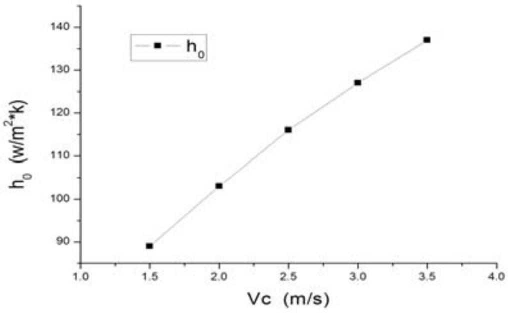

Fig. 2 Air-side heat transfer coefficient with the change of air velocity

When evaporation temperature is maintained 5 centigrade, alter air velocity and heat flux, the change of heat transfer coefficient is showed in Fig. 3. It indicates that heat transfer coefficient increases with the increase of air velocity and heat flux. But the scope of heat transfer coefficient's increase is less and less and air velocity's increase can bring yawp. So we should choose appropriate air velocity when evaporator is designed.

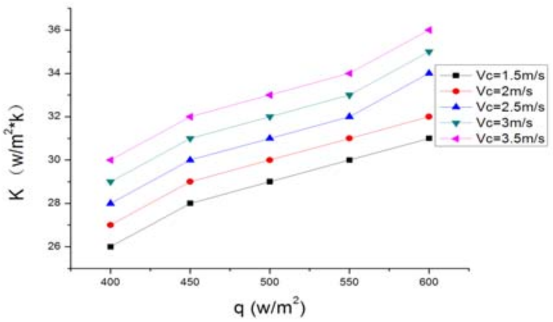

Fig. 3 The change of heat transfer coefficient in different air velocity and heat flux.

3.2 Impact of Evaporation Temperature

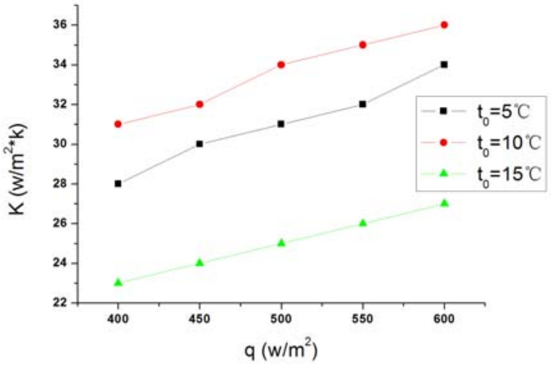

Fig. 4 The change of heat transfer coefficient in different evaporation temperature

Tab.3 Comparison of weather area in different air velocity

air velocity (m/s)	1.5	2.0	2.5	3.0	3.5
weather area (m^2)	0.084	0.081	0.077	0.074	0.071
reduce percent	13%	16%	20%	23%	26%

When air velocity is maintained 2.5m/s, alter evaporation temperature and heat flux, the change of heat transfer coefficient is showed in Fig.4. It indicates that heat transfer coefficient increases with the increase of heat flux.. When evaporation temperature changes from 5 to 10 centigrade, heat transfer coefficient also increases. But when evaporation temperature changes from 10 to 15 centigrade, heat transfer coefficient drops rapidly. The main reason is that the surface tension of refrigerant increases rapidly at 15 centigrade. The effect of surface tension is profound in microchannel. So increasing evaporation temperature appropriately can enhance heat transfer coefficient.

4. EPILOGUE

At present the development of boiling heat transfer is very rapid and the tube diameter is more and more small. But the results of all researchers are not unification. The impact of heat flux, tube diameter and flow form is not acknowledged. The conventional heat transfer correlations are difficult to be used in microchannel. Now there are no appropriate boiling heat transfer correlations, which can be used in microchannel. In the field of boiling heat transfer in microchannel, there are many difficults to be resolve. So it will be farther studied.

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